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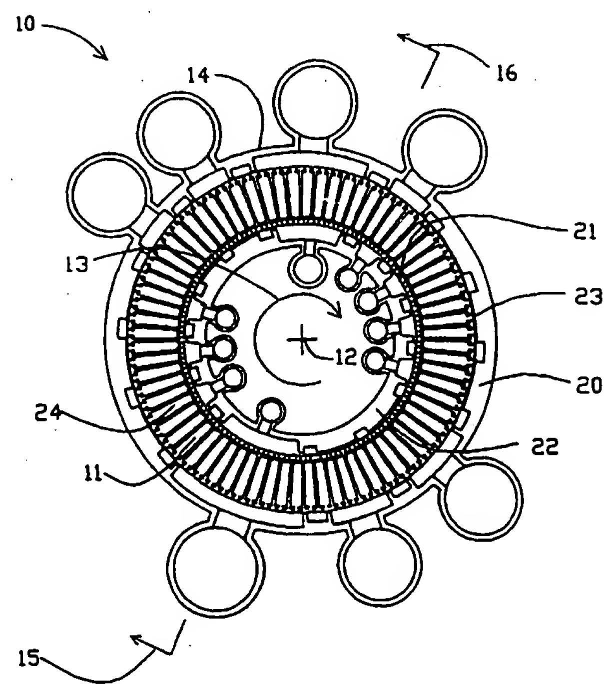
(54) Title: MODULAR PRESSURE SWING ADSORPTION APPARATUS

(57) Abstract

A rotary module for implementing a high frequency pressure swing adsorption process comprises a stator and a rotor rotatably coupled to the stator. The stator includes a first stator valve surface, a second stator valve surface, a plurality of first function compartments opening into the first stator valve surface, and a plurality of second function compartments opening into the second stator valve surface. The rotor includes a first rotor valve surface in communication with the first stator valve surface, a second rotor valve surface in communication with the second stator valve surface, and a plurality of flow paths for receiving adsorbent material therein. Each flow path includes a pair of opposite ends, and a plurality of apertures provided in the rotor valve surfaces and in communication with the flow path ends and the function ports for cyclically exposing each said flow path to a plurality of discrete pressure levels between the upper and lower pressures for maintaining uniform gas flow through the first and second function compartments.



MODULAR PRESSURE SWING ADSORPTION APPARATUS

FIELD OF THE INVENTION

The present invention relates to an apparatus for separating gas fractions from a gas mixture having multiple gas fractions. In particular, the present invention relates to a rotary valve gas separation system having a plurality of rotating adsorbent beds disposed therein for implementing a pressure swing adsorption process for separating out the gas fractions.

BACKGROUND OF THE INVENTION

Pressure swing adsorption (PSA) and vacuum pressure swing adsorption (VPSA) separate gas fractions from a gas mixture by coordinating pressure cycling and flow reversals over an adsorbent bed which preferentially adsorbs a more readily adsorbed component relative to a less readily adsorbed component of the mixture. The total pressure of the gas mixture in the adsorbent bed is elevated while the gas mixture is flowing through the adsorbent bed from a first end to a second end thereof, and is reduced while the gas mixture is flowing through the adsorbent from the second end back to the first end. As the PSA or VPSA cycle is repeated, the less readily adsorbed component is concentrated adjacent the second end of the adsorbent bed, while the more readily adsorbed component is concentrated adjacent the first end of the adsorbent bed. As a result, a "light" product (a gas fraction depleted in the more readily adsorbed component and enriched in the less readily adsorbed component) is delivered from the second end of the bed, and a "heavy" product (a gas fraction enriched in the more strongly adsorbed component) is exhausted from the first end of the bed.

The conventional system for implementing pressure swing adsorption or vacuum pressure swing adsorption uses two or more stationary adsorbent beds in parallel, with directional valving at each end of each adsorbent bed to connect the beds in alternating sequence to pressure sources and sinks. However, this system is often difficult and expensive to implement due to the complexity of the valving required.

Furthermore, the conventional PSA or VPSA system makes inefficient use of applied energy, because feed gas pressurization is provided by a compressor whose delivery pressure is the highest pressure of the cycle. In PSA, energy expended in compressing the feed gas used for pressurization is then dissipated in throttling over valves over the instantaneous pressure difference between the adsorber and the high pressure supply. Similarly, in VPSA, where the lower pressure of the cycle is established by a vacuum pump exhausting gas at that pressure, energy is dissipated in throttling over valves during countercurrent blowdown of adsorbers whose pressure is being reduced. A further energy dissipation in both systems occurs in throttling of light reflux gas used for purge, equalization, cocurrent blowdown and product pressurization or backfill steps.

Numerous attempts have been made at overcoming the deficiencies associated with the conventional PSA or VPSA system. For example, Siggelin (U.S. Patent No. 3,176,446), Mattia (U.S. Patent No. 4,452,612), Davidson and Lywood (U.S. Patent No. 4,758,253), Boudet et al (U.S. Patent No. 5,133,784), Petit et al (U.S. Patent No. 5,441,559) and Schartz (PCT publication WO 94/04249) disclose PSA devices using rotary distributor valves having rotors fitted with multiple angularly separated adsorbent beds. Ports communicating with the rotor-mounted adsorbent beds sweep past fixed ports for feed admission, product delivery and pressure equalization. However, these prior art rotary distributor

the open aperture(s) from the pressurization/blowdown compartment into the adsorber, which by the end of the pressurization/blowdown step has attained approximately the same pressure as the pressurization/blowdown compartment(s). Each pressurization/blowdown compartment is in communication with typically several adsorbers being pressurized (in differing angular and time phase) at any given time, so the pressure in that compartment and the pressurization flow to that compartment are substantially steady.

The flow path through the adsorbers may be radial or axial. If the adsorbers are configured for radial flow, the first valve surface would preferably be radially inward when the less strongly adsorbed gas fraction has much higher density that the more strongly adsorbed fraction, and the first valve surface would preferably be radially outward when the less strongly adsorbed gas fraction has much lower density than the more strongly adsorbed fraction. Hence, for hydrogen purification in a radial flow embodiment, the feed gas would preferably be admitted to (and the higher molecular weight impurity fraction as heavy product is exhausted from) the first valve surface at an outer radius, while the hydrogen as first product gas is delivered from the second valve surface.

The present invention also includes the alternatives of (1) layered or laminated thin sheet adsorbers and (2) the centrifugally stabilized fine particle granular adsorbers to enable operation at exceptionally high cycle frequency. PSA cycle frequencies to at least 100 cycles per minute are practicable within the present invention, and will enable process intensification so that high productivity can be realized from compact modules. Cycle frequencies more rapid than about 50 cycles per minute will be achieved preferably with the layered thin sheet adsorbers, with the flow path in flow channels tangential to and between adjacent pairs of adsorbent loaded sheets, to obtain lower frictional pressure drop at high frequency than granular adsorbent.

Preferably, the increments between adjacent pressure levels are sized so that the gas flows entering or exiting the module are substantially steady in both flow velocity and pressure. As a result, the module can be operated with centrifugal or axial flow compressors and expanders, for most favourable efficiency and capital cost economies of scale. To reduce throttling losses, it is also preferred that the function compartments are shaped to provide uniform gas flow through the flow paths and/or the valve surfaces include sealing strips having tapered portions for providing uniform gas flow through the flow paths.

Since the orifices providing the valving function are immediately adjacent to the ends of the flow paths, the dead volume associated with prior art distribution manifolds is substantially reduced. Also, since the compartments communicating with the first and second valve surfaces are external to the valving function, the compartments do not contribute to dead volume of the adsorbers. As a result, high frequency pressure/vacuum swing adsorption is possible.

Also, in contrast to prior art PSA devices whose pressure vessels are subject to pressure cycling and consequent fatigue loading, the pressure vessel of the present invention operates under substantially static stresses, because each of the compartments operates under steady pressure conditions. Mechanical stresses on the rotor and its bearings are relatively small, because only small frictional pressure drops (at most equal to the interval between adjacent intermediate pressures) apply in the flow direction, while transverse pressure gradients between the adsorber elements are also small owing to the large number of elements. These features are important, since pressure vessel fatigue is a major concern and limitation in the design of PSA systems, especially working with corrosive gases or hydrogen at higher pressure or higher cycle frequency.

Fig. 15 shows a VPSA apparatus with 4 modules;

Fig. 16 shows a PSA apparatus with 5 modules;

Fig. 17 shows a simplified schematic of a VPSA cycle for oxygen production, using a split stream air compressor, a split stream vacuum pump as the countercurrent blowdown exhauster, and a split stream light reflux expander powering a product oxygen compressor;

Fig. 18 shows a radial flow rotary PSA module;

Fig. 19 shows an axial flow rotary PSA module;

Fig. 20 shows a double axial flow rotary PSA module;

Fig. 21 shows the first valve face of the embodiment of Fig. 19;

Fig. 22 shows the second valve face of the embodiment of Fig. 19;

Fig. 23 shows an adsorber wheel configurations based on laminated adsorbent sheet adsorbers for the embodiment of Fig. 19;

Fig. 24 shows a multistage centrifugal compressor with impulse turbine expanders for the light reflux and countercurrent blowdown;

Fig. 25 shows the light reflux impulse turbine runner with four nozzles;

Fig. 26 is an unrolled view of the light reflux expander impulse turbine;

Fig. 27 is an unrolled view of the countercurrent blowdown expander impulse turbine;

Fig. 28 shows a split stream axial compressor with three stages; and

Fig. 29 shows a composite pellet with zeolite material coated on a high specific gravity inert core, for centrifugally stabilized granular adsorbers in radial flow embodiments.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Figs. 1, 2, 3 and 4

A rotary module 10 according to the invention is shown in Figs. 1, 2, 3 and 4. The module includes a rotor 11 revolving about axis 12 in the direction shown by arrow 13 within stator 14. Fig. 4 is an axial section of the module 10, defined by arrows 15 and 16 in Fig. 1. Fig. 1 is a cross-section of the module 10, defined by arrows 17 and 18 in Fig. 4. Fig. 2 is the sectional view of the rotor 11 repeated from Fig. 1, with the stator deleted for clarity. Fig. 3 is the sectional view of the stator 14 repeated from Fig. 1, with details of the rotor deleted for clarity.

In general, the apparatus of the invention may be configured for flow through the adsorber elements in the radial, axial or oblique conical directions relative to the rotor axis. For operation at high cycle frequency, radial flow has the advantage that the centripetal acceleration will lie parallel to the flow path for most favourable stabilization of buoyancy-driven free convection, as well as centrifugal clamping of granular adsorbent with uniform flow distribution. As shown in Fig. 2, the rotor 11 is of annular section, having concentrically to axis 12 an outer cylindrical wall 20 whose external surface is first valve surface 21, and an inner cylindrical wall 22 whose internal surface is second valve surface 23. The rotor has (in the plane of the section defined by arrows 15 and 16 in Fig. 4) a total of "N" radial flow adsorber elements 24. An adjacent pair of adsorber elements 25 and 26 are separated by partition 27 which is structurally and sealingly joined to outer wall 20 and inner wall 22. Adjacent adsorber elements 25 and 26 are angularly spaced relative to axis 12 by an angle of [360° / N].

Adsorber element 24 has a first end 30 defined by support screen 31 and a second end 32 defined by support screen 33. The adsorber may be provided as granular adsorbent, whose packing voidage defines a flow path contacting the adsorbent between the first and second ends of the adsorber.

simultaneously receiving the first feed gas from compartment 52, or from adsorbers receiving both the first and second feed gases.

A first light reflux exit compartment 72 communicates to first light reflux exit manifold 73, which is maintained at a first light reflux exit pressure, here substantially the higher working pressure less frictional pressure drops. A first cocurrent blowdown compartment 74 (which is actually the second light reflux exit compartment), communicates to second light reflux exit manifold 75, which is maintained at a first cocurrent blowdown pressure less than the higher working pressure. A second cocurrent blowdown compartment or third light reflux exit compartment 76 communicates to third light reflux exit manifold 77, which is maintained at a second cocurrent blowdown pressure less than the first cocurrent blowdown pressure. A third cocurrent blowdown compartment or fourth light reflux exit compartment 78 communicates to fourth light reflux exit manifold 79, which is maintained at a third cocurrent blowdown pressure less than the second cocurrent blowdown pressure.

A purge compartment 80 communicates to a fourth light reflux return manifold 81, which supplies the fourth light reflux gas which has been expanded from the third cocurrent blowdown pressure to substantially the lower working pressure with an allowance for frictional pressure drops. The ordering of light reflux pressurization steps is inverted from the ordering or light reflux exit or cocurrent blowdown steps, so as to maintain a desirable "last out - first in" stratification of light reflux gas packets. Hence a first light reflux pressurization compartment 82 communicates to a third light reflux return manifold 83, which supplies the third light reflux gas which has been expanded from the second cocurrent blowdown pressure to a first light reflux pressurization pressure greater than the lower working pressure. A second light reflux pressurization compartment 84 communicates to a second light reflux return manifold 85, which supplies the second light reflux gas which has been expanded from the first cocurrent blowdown pressure to a second light reflux pressurization pressure greater than the first light reflux pressurization pressure. Finally, a third light reflux pressurization compartment 86 communicates to a first light reflux return manifold 87, which supplies the first light reflux gas which has been expanded from approximately the higher pressure to a third light reflux pressurization pressure greater than the second light reflux pressurization pressure, and in this example less than the first feed pressurization pressure.

Additional details are shown in Fig. 4. Conduits 88 connect first compartment 60 to manifold 61, with multiple conduits providing for good axial flow distribution in compartment 60. Similarly, conduits 89 connect second compartment 80 to manifold 81. Stator 14 has base 90 with bearings 91 and 92. The annular rotor 11 is supported on end disc 93, whose shaft 94 is supported by bearings 91 and 92. Motor 95 is coupled to shaft 94 to drive rotor 11. The rotor could alternatively rotate as an annular drum, supported by rollers at several angular positions about its rim and also driven at its rim so that no shaft would be required. A rim drive could be provided by a ring gear attached to the rotor, or by a linear electromagnetic motor whose stator would engage an arc of the rim. Outer circumferential seals 96 seal the ends of outer strip seals 42 and the edges of first valve surface 21, while inner circumferential seals 97 seal the ends of inner strip seals 44 and the edges of second valve surface 23. Rotor 11 has access plug 98 between outer wall 20 and inner wall 22, which provides access for installation and removal of the adsorbent in adsorbers 24.

A further most important benefit of the invention in radial flow embodiments arises in purification of very low molecular weight gases such as hydrogen and helium to remove higher molecular weight impurities. Here, the light product is separated radially inward, while the heavy impurities are separated radially outward by the centrifugal PSA apparatus of the present invention. In

matrix (woven or non-woven) on which the adsorbent material (e.g. zeolite crystallites is supported by a suitable binder (e.g., clay, silicate or coke binders). Typical thickness of an adsorbent sheet may be about 100 microns. The sheets 115 are installed with spacers on one or both sides to establish flow channels between adjacent pairs of sheets. The flow channels define the flow path approximately in the radial direction between first end 30 and second end 32 of the flow path in each adsorber element. Typical channel height would be about 50% to 100% of the adsorbent sheet thickness.

The adsorbent sheets comprise a reinforcement material, preferably glass fibre, but alternatively metal foil or wire mesh, to which the adsorbent material is attached with a suitable binder. For air separation to produce enriched oxygen, typical adsorbents are X, A or chabazite type zeolites, typically exchanged with calcium or lithium cations. The zeolite crystals are bound with silica, clay and other binders within the adsorbent sheet matrix.

Satisfactory adsorbent sheets have been made by coating a slurry of zeolite crystals with binder constituents onto the reinforcement material, with successful examples including non-woven fiber glass scrims, woven metal fabrics, and expanded aluminum foils. Spacers are provided by printing or embossing the adsorbent sheet with a raised pattern, or by placing a fabricated spacer between adjacent pairs of adsorbent sheets. Alternative satisfactory spacers have been provided as woven metal screens, fiber glass scrims, and metal foils with etched flow channels in a photolithographic pattern.

Typical experimental sheet thicknesses have been 150 microns, with spacer heights in the range of 100 to 150 microns, and adsorber flow channel length approximately 20 cm. Using X type zeolites, excellent performance has been achieved in oxygen separation from air at PSA cycle frequencies in the range of 50 to 100 cycles per minute.

Figs. 6 and 7

Fig. 6 shows a typical PSA cycle according to the invention, while Fig. 7 shows a similar PSA cycle with heavy reflux recompression of a portion of the first product gas to provide a second feed gas to the process.

In Figs. 6 and 7, the vertical axis 150 indicates the working pressure in the adsorbers and the pressures in the first and second compartments. Pressure drops due to flow within the adsorber elements are neglected. The higher and lower working pressures are respectively indicated by dotted lines 151 and 152.

The horizontal axis 155 of Figs. 6 and 7 indicates time, with the PSA cycle period defined by the time interval between points 156 and 157. At times 156 and 157, the working pressure in a particular adsorber is pressure 158. Starting from time 156, the cycle for a particular adsorber (e.g. 24) begins as the first aperture 34 of that adsorber is opened to the first feed pressurization compartment 46, which is fed by first feed supply means 160 at the first intermediate feed pressure 161. The pressure in that adsorber rises from pressure 158 at time 157 to the first intermediate feed pressure 161. Proceeding ahead, first aperture passes over a seal strip, first closing adsorber 24 to compartment 46 and then opening it to second feed pressurization compartment 50 which is feed by second feed supply means 162 at the second intermediate feed pressure 163. The adsorber pressure rises to the second intermediate feed pressure.

First aperture 34 of adsorber 24 is opened next to first feed compartment 52, which is maintained at substantially the higher pressure by a third feed supply means 165. Once the adsorber pressure has risen to substantially the higher working pressure, its second aperture 35 (which has been closed to all

The adsorber is then repressurized by light reflux gas after the first and second apertures close to compartments 60 and 80. In succession, while the first aperture 34 remains closed at least initially, (a) the second aperture 35 is opened to first light reflux pressurization compartment 82 to raise the adsorber pressure to the first light reflux pressurization pressure 190 while receiving third light reflux gas from the third light reflux pressure letdown means 174, (b) the second aperture 35 is opened to second light reflux pressurization compartment 84 to raise the adsorber pressure to the second light reflux pressurization pressure 191 while receiving second light reflux gas from the second light reflux pressure letdown means 172, and (c) the second aperture 35 is opened to third light reflux pressurization compartment 86 to raise the adsorber pressure to the third light reflux pressurization pressure 192 while receiving first light reflux gas from the first light reflux pressure letdown means 170. Unless feed pressurization has already been started while light reflux return for light reflux pressurization is still underway, the process (as based on Figs. 6 and 7) begins feed pressurization for the next cycle after time 157 as soon as the third light reflux pressurization step has been concluded.

The pressure variation waveform in each adsorber would be a rectangular staircase if there were no throttling in the first and second valves. In order to provide balanced performance of the adsorbers, preferably all of the apertures are closely identical to each other.

The rate of pressure change in each pressurization or blowdown step will be restricted by throttling in ports (or in clearance or labyrinth sealing gaps) of the first and second valve means, or by throttling in the apertures at first and second ends of the adsorbers, resulting in the typical pressure waveform depicted in Figs. 6 and 7. Alternatively, the apertures may be opened slowly by the seal strips, to provide flow restriction throttling between the apertures and the seal strips, which may have a serrated edge (e.g. with notches or tapered slits in the edge of the seal strip) so that the apertures are only opened to full flow gradually. Excessively rapid rates of pressure change would subject the adsorber to mechanical stress, while also causing flow transients which would tend to increase axial dispersion of the concentration wavefront in the adsorber. Pulsations of flow and pressure are minimized by having a plurality of adsorbers simultaneously transiting each step of the cycle, and by providing enough volume in the function compartments and associated manifolds so that they act effectively as surge absorbers between the compression machinery and the first and second valve means.

It will be evident that the cycle could be generalized by having more or fewer intermediate stages in each major step of feed pressurization, countercurrent blowdown exhaust, or light reflux. Furthermore, in air separation or air purification applications, a stage of feed pressurization (typically the first stage) could be performed by equalization with atmosphere as an intermediate pressure of the cycle. Similarly, a stage of countercurrent blowdown could be performed by equalization with atmosphere as an intermediate pressure of the cycle.

Fig.8

Figs. 8 - 15 are simplified schematics of PSA systems using the module 10 of Figs. 1 - 4 as the basic building block, and showing the connections from the first and second manifolds of the module to machinery for compression and expansion of gases in typical applications. In Figs. 8 - 15, the reference numerals of the first and second manifolds are as defined for Fig. 3.

Fig. 8 is a simplified schematic of a PSA system for separating oxygen from air, using nitrogen-selective zeolite adsorbents. The light product is concentrated oxygen, while the heavy product is nitrogen-enriched air usually vented as waste. The cycle lower pressure 152 is nominally atmospheric pressure. Feed air is introduced through filter intake 200 to a feed compressor 201. The feed compressor

While heaters 225 and 241 are means to provide heat to the expanders, intercoolers 203, 205 and 207 are means to remove heat from the feed compressor and serve to reduce the required power of the higher compressor stages. The haters and intercoolers are optional features of the invention.

If light reflux heater 249 operates at a sufficiently high temperature so that the exit temperature of the light reflux expansion stages is higher than the temperature at which feed gas is delivered to the feed manifolds by conduits 212, 214 and 216, the temperature of the second ends 35 of the adsorbers 24 may be higher than the temperature of their first ends 34. Hence, the adsorbers have a thermal gradient along the flow path, with higher temperature at their second end relative to the first end. This is an extension of the principle of "thermally coupled pressure swing adsorption" (TCPSA), introduced by Keefer in U.S. Patent No. 4,702,903. Adsorber rotor 11 then acts as a thermal rotary regenerator, as in regenerative gas turbine engines having a compressor 201 and an expander 220. Heat provided to the PSA process by heater 225 assists powering the process according to a regenerative thermodynamic power cycle, similar to advanced regenerative gas turbine engines approximately realizing the Ericsson thermodynamic cycle with intercooling on the compression side and interstage heating on the expansion side.

In the instance of PSA applied to oxygen separation from air, the total light reflux flow is much less than the feed flow because of the strong bulk adsorption of nitrogen. Accordingly the power recoverable from the expanders is much less than the power required by the compressor, but will still contribute significantly to enhanced efficiency of oxygen production. By operating the adsorbers at moderately elevated temperature and using strongly nitrogen-selective adsorbents such as Ca-X, Li-X or calcium chabazite zeolites, a PSA oxygen generation system can operate with favourable performance and exceptional efficiency. While higher temperature of the adsorbent will reduce nitrogen uptake and selectivity, the isotherms will be more linear. Effective working capacity in superatmospheric pressure PSA cycles may be enhanced by operation in TCPSA mode with an elevated temperature gradient in the adsorbers. Working with adsorbents such as Ca-X and Li-X, recent conventional practice has been to operate ambient temperature PSA at subatmospheric lower pressures in so-called "vacuum swing adsorption" (VSA), so that the highly selective adsorbents operate well below saturation in nitrogen uptake, and have a large working capacity in a relatively linear isotherm range. At higher temperatures, saturation in nitrogen uptake is shifted to more elevated pressures, so the optimum PSA cycle higher and lower pressures are also shifted upward. For satisfactory operation of the apparatus of Fig. 8, the typical operating temperature of the second ends of the adsorbers may be approximately 50°C for Ca-X or Li-X, and 100° to 150°C for calcium chabazite.

If high energy efficiency were not of highest importance, the light reflux expander stages and the countercurrent blowdown expander stages could be replaced by restrictor orifices or throttle valves for pressure letdown, as illustrated in Fig. 10. The schematic of Fig. 8 shows a single shaft supporting the compressor stages, the countercurrent blowdown or exhaust expander stages, and the light reflux stages, as well as coupling the compressor to the prime mover. However, it should be understood that separate shafts and even separate prime movers may be used for the distinct compression and expansion stages within the scope of the present invention.

It should also be understood that the number of compression stages and the number of expansion stages (as well as the number of vacuum pump stages in the embodiment of Fig. 9 below) may be varied within the scope of the invention. Generally and for equal stage efficiency of the compressor or expander type chosen, a larger number of stages will improve the PSA process efficiency, since the irreversible equalization expansions over the first and second orifices will be performed over narrower

For hydrogen duty, positive displacement expansion and compression machinery (e.g. twin screw machines) may be preferred because of the low molecular weight of the gas. Such machines may be adapted in accordance with the invention with extra inlet and/or discharge ports to accept and deliver gas at multiple intermediate pressures.

Performance and productivity of PSA hydrogen recovery from refinery offgases (with the adsorbers working at near ambient temperature) will be greatly enhanced by operating with the lower working pressure as low as possible and preferably near atmospheric pressure. However, the tail gas is usually delivered at a pressure of at least 5 or 6 atmospheres, for disposal to the refinery fuel gas header. Compression costs, particularly for combustible gases under refinery safety constraints, may be prohibitively high.

The apparatus of Fig. 11 is configured to accept first and second feed gas mixtures, the first having a higher concentration of the less readily adsorbed component (e.g. hydrogen) while the second is more concentrated than the first feed gas mixture in the more readily adsorbed fraction. The first feed gas is supplied at substantially the higher working pressure by first infeed conduit 280 to first feed manifold 53, while the second feed gas is supplied at substantially the higher working pressure by second infeed conduit 281 to first feed manifold 55. Each adsorber receives the second feed gas after receiving the first feed gas, so that the concentration profile in the adsorber is monotonically declining in concentration of the more readily adsorbed component along its flow path from first end 34 to second end 35 of the flow path in adsorber element 24. All but the final pressurization steps are here achieved with light reflux gas. The final feed pressurization (from the third light reflux pressurization pressure 192 directly to the higher pressure 151) is achieved as the first end of each adsorber is opened to compartment 52 communicating to manifold 53. Additional feed pressurization steps could readily be incorporated as in the embodiment of Fig. 8.

In this embodiment, the tail gas (heavy product) is discharged from second product delivery conduit at a higher pressure than the lower working pressure, in this example being approximately the first countercurrent blowdown pressure 180 of Fig. 6 with conduit 240 being first exhaust means 181. Tail gas is recompressed by tail gas compressor 290, with compressor first stage 291 being the third exhaust means 184 compressing the first product gas from exhaust manifold 61 via conduit 246, and delivering the first product gas after first stage compression through intercooler 292 to compressor second stage 293 which itself is the second exhaust means compressing second countercurrent blowdown gas from manifold 59 via conduit 244.

Fig. 12

Fig. 12 shows a PSA apparatus with heavy reflux to obtain either higher enrichment and purity of the more readily adsorbed component into the heavy product, or higher yield (recovery) of the less readily adsorbed component into the light product. This apparatus may also be configured to deliver the heavy product at elevated pressure, here approximately the higher working pressure so that both product gases are delivered at about the higher pressure.

The apparatus of Fig. 12 has infeed conduit 300 to introduce the feed gas at substantially the higher pressure to first feed manifold 53. As in the example of Fig. 11, adsorber pressurization is achieved mainly by light reflux, with a final feed pressurization step through manifold 53.

A multistage heavy reflux compressor 301 has a first stage 302 as third exhaust means 184 of Fig. 7, drawing heavy gas by conduit 246 from first product exhaust manifold 61, and compressing this gas through intercooler 303 to second stage 304. Heavy reflux compressor second stage 304 as second

expander 221 and light reflux expander 220 through shaft 222. This configuration enables operation of a motor driven feed compressor with a limited number of stages (here 2 stages) to operate a PSA cycle with a larger number of feed supply pressures (here the three pressures 161, 163 and 151 of Fig. 6), since the free rotor compressor has dual functions as means to boost feed pressure by application of thermally boosted expansion energy recovery, and means to split the stage intermediate pressures for supply to the PSA module.

Fig. 15

Fig. 15 shows a VPSA oxygen generation plant with 4 modules in parallel, each having a free rotor booster compressor powered by energy recovery expanders, and the entire apparatus having a single prime mover 350 which may for example be an electric motor or a gas turbine. Prime mover 350 drives first feed compressor 351 on shaft 352. Feed compressor 351 has a first stage 353 drawing feed gas from infeed conduit 200, and a third stage 354. The second stage of feed compression is provided by the free rotor compressors of each module. The first feed compressor 351 in this embodiment also includes an exhaust vacuum pump 355 likewise coupled to shaft 352.

The plant includes four identical modules 10A, 10B, 10C and 10D. In Figs. 15 and 16, component nomenclature and reference numerals follow that established for Figs. 1 - 14, with a suffix A to D appended to the reference numerals for module components, and each component so identified with reference to any one module will be identically found in each of the other modules. The first manifolds are identified with reference to module 10D as 48D and 51D for feed pressurization, 53D for feed supply at the higher pressure, 57D and 59D for countercurrent blowdown, and 61D for exhaust at the lower pressure. The second manifolds are identified with reference to module 10C as 71C communicating to light product delivery manifold 360 and delivery conduit 218, light reflux exit manifolds 73C, 75C, 77C and 79C, and light reflux return manifolds 81C, 83C, 85C and 87C.

The identical free rotor compressor for each module will be described with reference to module 10B. Free rotor compressor assembly 370B includes feed compression second stage 371B and vacuum pump 372B, both coupled by shaft 373B to light reflux expander 220B. Feed gas compressed by feed compressor first stage 353 is conveyed by feed manifold 376 in parallel to the first feed pressurization manifold (e.g. 48D) of each module, and to the inlet of feed compression second stage (e.g. 371B) of the free rotor compressor assembly (e.g. 370B) of each module which delivers further compressed feed pressurization gas to the second feed pressurization manifold (e.g. 51D) of each module. Feed gas compressed to the higher pressure by third feed compressor stage 354 is conveyed by feed manifold 377 in parallel to the first feed supply manifold (e.g. 53D) of each of the modules. Heavy gas at the lower pressure is drawn from the heavy compartment (e.g. 61D) of each module through vacuum exhaust manifold 378 to exhaust vacuum pump 355 as the third exhaust means. Countercurrent blowdown gas from the first countercurrent blowdown manifold (e.g. 57D) of each module is discharged by e.g. conduit 240B as first exhaust means, while countercurrent blowdown gas from the second countercurrent blowdown manifold (e.g. 59D) of each module is exhausted by vacuum pump (e.g. 372B) of the free rotor compressor assembly as second exhaust means, delivering the heavy tail gas to the module heavy product or waste gas exhaust, e.g. 243B.

Fig. 16

Fig. 16 shows a PSA apparatus with 5 modules 10A - 10E. In this embodiment, the prime mover, all compressor stages and all expander stages are directly mechanically coupled (e.g. on a single

Rotor 11 is driven by motor 95 connected to stub shaft 511 by shaft 94 penetrating housing 513 through shaft seal 522. First end plate 510 has no perforations that might compromise purity of the light product gas by leakage from the first valve surface to the second valve surface. Second end plate 515 is penetrated at bushing 530 by the second valve stator. Second valve stator 41 is a stationary pintle within rotor 11, with guide bushings 530 and 532, and is attached to the second bearing housing 518 at assembly face 534. Bearings 512 and 517 may be much smaller in diameter than the outer diameter of rotor 11 at sealing face 21. A shaft seal 535 is provided between shaft 516 and bearing 517, to prevent contamination of the light product gas by leakage from chamber 536 adjacent the first valve sealing face 21 to chamber 537 adjacent the second valve sealing face 23.

Preferably, seal 535 is tight against leakage so that product purity is not compromised. By configuring this seal at smaller diameter than the valve sealing faces, frictional torque from shaft seal 535 is greatly reduced than if this seal were at the full rotor diameter. Leakage across seals in the first valve face is much less important, because moderate leakage across those seals simply reduced the volumetric efficiency of the process. Similarly, moderate leakage across the seals in the second valve face may be tolerated, as the concentration of light reflux gases and the light product gas that may leak across those seals is almost identical. Because moderate leakage across seals in the first valve surface (including circumferential seals 96), and across seals in the second valve surface (including circumferential seals 97), can be accepted, all of those seals may be designed for relatively light mechanical engagement to minimize frictional torque. In fact, use of narrow gap clearance seals or labyrinth seals with zero mechanical rubbing friction is an attractive option especially for larger capacity modules operating at high cycle frequency (e.g. 50 or 100 cycles per minute) where seal leakage flows would have a minimal effect on overall efficiency. Preferably, the seals in the first and second valve faces have consistent performance and leakage, so that all "N" adsorbers experience the same PSA cycle flow and pressure regime as closely as possible, without being upset by variations in leakage between the adsorbers.

Hence an important benefit of the present invention is that close tolerance sealing is only required on one dynamic rotary seal, shaft seal 535, whose diameter has been made much smaller than the rotor diameter to reduce the sealing perimeter as well as mechanical friction power loss. For a given rotary seal section and loading, rubbing friction power loss at given RPM is proportional to the square of the sealing face diameter.

Because of the compactness (similar to an automotive turbocharger) of a "turbocompressor" oxygen booster as described for Fig. 17above, it is possible to install a split stream light reflux expander 220 with close-coupled light product compressor 396 inside the light valve stator. Compressed oxygen product is delivered by conduit 218.

<u>Fig.19</u>

Fig. 19 shows an axial flow rotary PSA module 600 for smaller scale oxygen generation. The flow path in adsorbers 24 is now parallel to axis 601. A better understanding will be obtained from Figs. 20, 21 and 22, which are cross sections of module 600 in the planes respectively defined by arrows 602 - 603, 604 – 605, and 606 - 607. Fig. 19 is an axial section of module 600 through compartments 54 and 70 at the higher pressure, and compartments 60 and 80 at the lower pressure. The adsorber rotor 11 contains the "N" adsorbers 24 in adsorber wheel 608, and revolves between the first valve stator 40 and the second valve stator 41. Compressed feed air is supplied to compartment 54 as indicated by arrow 501, while nitrogen enriched exhaust gas is exhausted from compartment 60 as indicated by arrow 502.

Typical closed sector 675 provides a transition for an adsorber, between being open to compartment 56 and open to compartment 58. Gradual opening is provided at the leading edges 677 and 678 of compartments, so as to achieve gentle pressure equalization of an adsorber being opened to a new compartment. Much wider closed sectors (e.g. 676) are provided to substantially close flow to or from one end of the adsorbers when pressurization or blowdown is being performed from the other end.

Sealing between compartments at typical closed sectors (e.g. 675) may be provided by rubbing seals on either stator or rotor against a ported hard-faced sealing counter face on the opposing rotor or stator, or by narrow gap clearance seals on the stator with the area of the narrow sealing gap defined by the cross hatched area of the nominally closed surface. Rubbing seals may be provided as radial strip seals, with a self-lubricating solid material such as suitable PTFE compounds or graphite, or as brush seals in which a tightly packed brush of compliant fibers rubs against the counter face.

If the rubbing seals are on the rotor (between adjacent adsorbers), cross-hatched sectors 675 and 676 would be non-ported portions of the hard-faced sealing counter face on the stator. If the rubbing seals are on the stator, the ported hard-faced counter face is on the rotor valve face. Those rubbing seals could be provided as full sector strips for narrow closed sectors (e.g. 675). For the wider closed sectors (e.g. 676), narrow radial rubbing seals may be used as the edges 678 and 679, and at intervals between those edges, to reduce friction in comparison with rubbing engagement across the full area of such wide sectors.

Clearance seals are attractive, especially for larger scale modules with a very large number "N" of adsorbers in parallel. The leakage discharge coefficient to or from the clearance gap varies according to the angular position of the adsorber, thus providing gentle pressure equalization as desired. The clearance gap geometry is optimized in typical nominally closed sectors (e.g. 675) so that the leakage in the clearance gap is mostly used for adsorber pressure equalization, thus minimizing through leakage between compartments. The clearance gap may be tapered in such sectors 675 to widen the gap toward compartments being opened, so that the rate of pressure change in pressure equalization is close to linear. For wide closed sectors (e.g. 676) the clearance gap would be relatively narrow as desired to minimize flows at that end of adsorbers passing through those sectors.

For all types of valve face seals described above, it is preferable that consistent performance be achieved over time, and that all "N" adsorbers experience the same flow pattern after all perturbations from seal imperfections. This consideration favours placing rubbing seals on the stator so that any imperfections are experienced similarly by all adsorbers. If the seals are mounted on the rotor between adsorbers, it is preferable that they are closely identical and highly reliable to avoid upsetting leakages between adjacent adsorbers.

To compensate for misalignment, thermal distortion, structural deflections and wear of seals and bearings, the sealing system should have a suitable self-aligning suspension. Thus, rubbing seal or clearance seal elements may be supported on elastomeric supports, bellows or diaphragms to provide the self-aligning suspension with static sealing behind the dynamic seal elements. Rubbing seals may be energized into sealing contact by a combination of elastic preload and gas pressure loading.

Clearance seals require extremely accurate gap control, which may be established by rubbing guides. Clearance seal gap control may also be achieved by a passive suspension in which the correct gap is maintained by a balance between gas pressure in the gap of a clearance seal segment, and the pressures of adjacent compartments loading the suspension behind that segment. For seal elements between blowdown compartments, a simple passive self-adjusting suspension should be stable. Active control elements could also be used to adjust the clearance seal gap, with feedback from direct gap height

For particular advantage in smaller plant capacities, considerable simplification is obtained in the embodiment of Figs. 24 - 27 by using partial admission impulse turbines for countercurrent blowdown and light reflux expansion, with each expander stage occupying a sectoral arc of the corresponding turbine on a single runner wheel. This approach is practicable because the stages for each turbine expand gases of approximately similar composition across adjacent pressure intervals.

Fig. 25 is a section of Fig. 24, defined by arrows 451 and 452, across the plane of light reflux impulse turbine runner 416. Fig. 24 is a section of Fig. 25, in the plane indicated by arrows 453 and 454. Runner 416 rotates about axis 406 in the direction indicated by arrow 455. Runner 416 has blades 456 mounted on its rim. Fig. 26 is a projected view of the light reflux expander impulse turbine, unrolled around 360° of the perimeter of the impulse turbine as indicated by the broken circle 458 with ends 459 and 460 in Fig. 25.

The light reflux turbine has four nozzles serving the four 90° quadrants of the runner to provide the four expansion stages, including first nozzle 461 receiving flow from port 462 communicating to conduit 224, second nozzle 463 receiving flow from port 464 communicating to conduit 228, third nozzle 465 receiving flow from port 466 communicating to conduit 232, and fourth nozzle 467 receiving flow from port 468 communicating to conduit 236.

The first stage light reflux flow from nozzle 461 impinges blades 456, and is collected in diffuser 471 and discharged at the reduced pressure by port 472 communicating to conduit 227. Similarly the light reflux flow from nozzle 463 is collected in diffuser 473 and flows by port 474 to conduit 231, the light reflux flow from nozzle 465 is collected in diffuser 475 and flows by port 476 to conduit 235, and the light reflux flow from nozzle 467 is collected in diffuser 477 and flows by port 478 to conduit 239. To minimize interstage leakage losses, the channel gap 479 between the casing 403 and blades 456 of runner 416 is appropriately narrow between quadrants.

The exhaust expander turbine, or countercurrent blowdown expander turbine, has two stages. Its sectional arrangement is similar to that depicted in Fig. 25, except that two rather than four nozzles and diffusers are required for the two exhaust stages. Fig. 27 is an unrolled projection around exhaust turbine runner 415 as indicated by broken circle 458 for the light reflux turbine. The exhaust turbine has impulse blades 480 on runner 415. Nozzle 481 receives the first countercurrent blowdown stream by port 482 communicating to conduit 240, while nozzle 483 receives the second countercurrent blowdown stream by port 484 communicating to conduit 244. Nozzles 481 and 483 have guide vanes 485 and 486, and direct the countercurrent blowdown flows to impinge on blades 480 in opposite half sectors of the turbine 415. After deflection by blades 480, the expanded flow from nozzle 481 is collected in diffuser 491, and is passed to collector ring manifold 492. The flow from nozzle 483 likewise passes the blades 480 and is collected in diffuser 493 joining manifold 492 to deliver the combined low pressure exhaust flow by exhaust port 494 which is connected to the discharge 243.

Fig. 28

Fig. 28 shows a three stage axial flow split stream compressor 700. While it is known in the prior art to divert minor bleed flows between stages of multistage axial flow compressors or expanders, compressor 700 has nested annular diffusers for splitting fractionally large intermediate flows from the main flow between stages.

Compressor 700 may represent split stream compressor 201 of Fig. 4, and has a scroll housing 701 with feed inlet 391, first discharge port 392, second discharge port 393 and third discharge port 394. Rotor 702 is supported by bearings 703 and 704 with shaft seals 705 and 706 within housing 701, and is

acceleration of gravity at the outer radius. The adsorbent beds, within the rotor and closer to the axis, are subject to a much smaller centripetal acceleration.

Ballasted composite pellet 800 has an inert core 801 of a dense material, surrounded by a coating 802 of macroporous zeolite material similar to the material of conventional adsorbent pellets. The core material may be selected for high density, high heat capacity, high thermal conductivity and compatibility for adhesion to zeolite binders as well as for thermal expansion. Suitable core materials include transition metal oxides, most simply iron oxide, as well as solid iron or nickel-iron alloys.

If the diameter of core 801 is e.g. 790 microns, and the radial thickness of coating 802 is e.g. 105 microns so that the overall diameter of a spherical pellet 800 is 1 mm, the volume of the pellet is then 50% inert and 50% active macroporous adsorbent. In a packed bed using such composite pellets, the active volume of adsorbent has been reduced by 50%, while the fractional bed voidage of the active material has been increased from the typical 35% of spherical granular media to approximately 50%. This might seem to be an inferior packed bed, with half as much useful material and reduced effective selectivity performance because of the high effective void fraction. Unexpectedly, this can be a superior packed bed, because pressure drop and mass transfer resistance are both reduced, so that the PSA cycle can be operated at higher cycle frequency without excessive pressure drop and without risk of fluidization. At the same cycle frequency, pressure drops are reduced by the smaller flows in proportion to the smaller active adsorbent inventory for the same voidage channels, while mass transfer through the macropores only has to take place through a relatively thin shell. The inert material also acts as thermal ballast to isothermalize the adsorber against thermal swings due to heat of adsorption.

While the higher void fraction will reduce product yield at specified purity in the uneconomic regime of very low cycle frequency, product yield and productivity are actually enhanced in the economic regime of higher cycle frequency. Degradations of product yield and process energy efficiency (at specified product purity) will result from mass transfer resistance and pressure drop, and those degradations are more severe for the conventional bed than for the present inventive granular adsorber of composite pellets.

Such composite pellets are very useful in the radial flow embodiment of the rotary adsorber module, since the heavy composite pellets are centrifugally stabilized very positively, even as mass transfer resistance and pressure drop are reduced. Such composite pellets will also be very useful in axial flow embodiments, as well as non-rotary adsorbers, with vertically oriented flow path. Again, cycle frequency can be increased, while performance can be enhanced in terms of productivity, yield and efficiency at the most economic operating point. Consider Figs. 4 and 18 to be vertical views of radial flow embodiments. The vertical axis embodiment of Fig. 4 will benefit from centrifugal stabilization if its rotor radius and cycle frequency are high enough. The horizontal axis embodiment of Fig. 18 will have centripetal acceleration assisting the gravitational field to suppress fluidization in the feed production step with upward flow from compartment 54 to compartment 70 at higher pressure, while the centripetal acceleration will assist pressure drop in the purge step with upward flow from compartment 80 to compartment 60 at lower pressure to prevent downward collapse of the adsorbers at the top of their rotational orbit. The adsorbent beds are supported at their first end (radially outside) by a first set of screens, and retained against collapsing when the rotor is stopped by a second set of screens at their second end (radially inside). Hence, the adsorbent beds are centrifugally clamped on the first screens by centripetal acceleration with the rotor acting as a centrifuge.

While composite pellets 800 are shown in Fig. 29 as spherical, other geometries are also attractive. For example, cylindrical composite pellets might be made by dip-coating the zeolite and

WE CLAIM:

1. A rotary module for implementing a pressure swing adsorption process having an operating pressure cycling between an upper pressure and a lower pressure for extracting a first gas fraction and a second gas fraction from a gas mixture including the first and second fractions, the rotary module comprising:

a stator including a first stator valve surface, a second stator valve surface, a plurality of first function compartments opening into the first stator valve surface, and a plurality of second function compartments opening into the second stator valve surface; and

a rotor rotatably coupled to the stator and including a first rotor valve surface in communication with the first stator valve surface, a second rotor valve surface in communication with the second stator valve surface, a plurality of flow paths for receiving adsorbent material therein, each said flow path including a pair of opposite ends, and a plurality of apertures provided in the rotor valve surfaces and in communication with the flow path ends and the function ports for cyclically exposing each said flow path to a plurality of discrete pressure levels between the upper and lower pressures for maintaining uniform gas flow through the first and second function compartments.

2. The rotary module according to claim 1, wherein the function compartments are shaped to provide uniform gas flow through the flow paths.

3. The rotary module according to claim 1, wherein at least one of the valve surfaces includes a sealing strip for reducing gas flow loss between the valve surfaces, the sealing strip including a tapered portion for providing uniform gas flow through the flow paths.

4. The rotary module according to claim 3, wherein the sealing strip is shaped to provide rapid closing of the flow paths.

5. The rotary module according to claim 1, wherein each said function compartment simultaneously communicates with at least two flow paths for providing uniform gas flow through the function compartments.

6. The rotary module according to claim 1, wherein each said function compartment is coupled immediately adjacent to a respective end of a respective one of the flow paths for implementing high frequency pressure swing adsorption.

7. The rotary module according to claim 6, wherein the function compartments are positioned a distance from the respective flow path ends sufficient for implementing the pressure swing adsorption process at a rotor rotational speed of at least 20 revolutions per minute.

8. The rotary module according to claim 1, wherein the function compartments include a plurality of pressurization compartments for subjecting the flow paths to a plurality of incremental pressures increases.

18. The rotary module according to claim 1, wherein the function compartments are disposed around the respective valve surfaces for conveying gas along the flow paths in a common predetermined sequence for each flow path, the sequence for each flow path comprising delivering the gas mixture at the upper pressure from a gas feed function compartment to the flow path end adjacent the first rotor valve surface while removing light product gas at the upper pressure from the flow path end adjacent the second rotor valve surface to a light product function compartment, removing heavy product gas at the lower pressure from the flow path end adjacent the first rotor valve surface to a heavy product gas function compartment, and delivering gas at a pressure intermediate the upper and lower pressure from a repressurization function compartment to the flow path end adjacent the first rotor valve surface ahead of the gas feed function compartment.

19. The rotary module according to claim 1, wherein the function compartments are disposed around the respective valve surfaces for conveying gas along the flow paths in a common predetermined sequence for each flow path, the sequence for each flow path comprising delivering the gas mixture at the upper pressure from a gas feed function compartment to the flow path end adjacent the first rotor valve surface while removing light product gas at the upper pressure from the flow path end adjacent the second rotor valve surface to a light product function compartment, removing gas at a pressure intermediate the upper and lower pressures from the flow path end adjacent the second rotor valve surface to a cocurrent blowdown function compartment, and removing heavy product gas at the lower pressure from the flow path end adjacent the first rotor valve surface to a heavy product gas function compartment.

20. The rotary module according to claim 1, wherein the function compartments are disposed around the respective valve surfaces for conveying gas along the flow paths in a common predetermined sequence for each flow path, the sequence for each flow path comprising delivering the gas mixture at the upper pressure from a gas feed function compartment to the flow path end adjacent the first rotor valve surface while removing light product gas at the upper pressure from the flow path end adjacent the second rotor valve surface to a light product function compartment, removing gas at a pressure intermediate the upper and lower pressures from the flow path end adjacent the first rotor valve surface to a countercurrent blowdown function compartment, and removing heavy product gas at the lower pressure from the flow path end adjacent the first rotor valve surface to a heavy product gas function compartment.

21. The rotary module according to claim 1, wherein each said flow path includes a laminated sheet adsorber.

22. A rotor module for use with a stator for implementing a pressure swing adsorption process having an operating pressure cycling between an upper pressure and a lower pressure, the rotor module comprising:

an annular rotor including a first rotor valve surface for communicating with a first stator valve surface of the stator, a second rotor valve surface for communicating with a second stator valve surface of the stator, a plurality of flow paths spaced around the rotor and extending between the rotor valve surfaces, and a plurality of apertures provided in the rotor valve surfaces in communication with the flow paths for cyclically exposing each said flow path to a plurality of discrete pressure levels between the

therein, each said flow path including a pair of opposite ends, and a plurality of apertures provided in the rotor valve surfaces and in communication with the flow path ends and the function ports; and

compression/expansion machinery coupled to the rotary module for maintaining the function ports at a plurality of discrete pressure levels between an upper pressure and a lower pressure for maintaining uniform gas flow through the first and second function compartments.

33. The pressure swing adsorption system according to claim 32, wherein the function compartments include a plurality of gas feed compartments, and the compression/expansion machinery comprises a multi-stage compressor including a plurality of pressure output ports, each said pressure output port being coupled to a respective one of the feed compartments for delivering feed gas to the flow paths at a plurality of pressure increments.

34. The pressure swing adsorption system according to claim 33, wherein the multi-stage compressor comprises a centrifugal compressor having a plurality of stages, each said stage including a gas inlet, a diffuser, and an impeller coupled to the gas inlet and having an axis of rotation for accelerating gas from the gas inlet towards the diffuser.

35. The pressure swing adsorption system according to claim 33, wherein the multi-stage compressor comprises a multi-stage axial flow split stream compressor including a plurality of annular stator rings of progressively decreasing diameter, each said stator ring including an annular flow area and a plurality of stator blades, and a rotor having an axis of rotation and including a plurality of rotor blades cooperating with the stator blades for compressing gas flow through the flow area, at least one of the said stator rings further including a collector and a diffuser for apportioning the compressed gas flow between the collector and the flow area of a subsequent one of the stator rings.

36. The pressure swing adsorption system according to claim 33, wherein the function compartments include a plurality of blowdown compartments, and the compression/expansion machinery includes a multi-stage vacuum pump coupled to the compressor, the vacuum pump including a plurality of pressure inlet ports, each said pressure inlet port being coupled to a respective one of the blowdown compartments for receiving blowdown gas from the flow paths at a plurality of pressure increments.

37. The pressure swing adsorption system according to claim 33, wherein the function compartments include a plurality of blowdown compartments, and the pressure swing adsorption system includes a plurality of throttle orifices coupled to the blowdown compartments for releasing blowdown gas from the flow paths at a plurality of pressure increments.

38. The pressure swing adsorption system according to claim 32, wherein the function compartments include a plurality of countercurrent blowdown compartments and a plurality of cocurrent blowdown compartments, and the compression/expansion machinery comprises a first expander coupled to the countercurrent blowdown compartments and a second expander coupled to the first expander and to the cocurrent blowdown compartments.

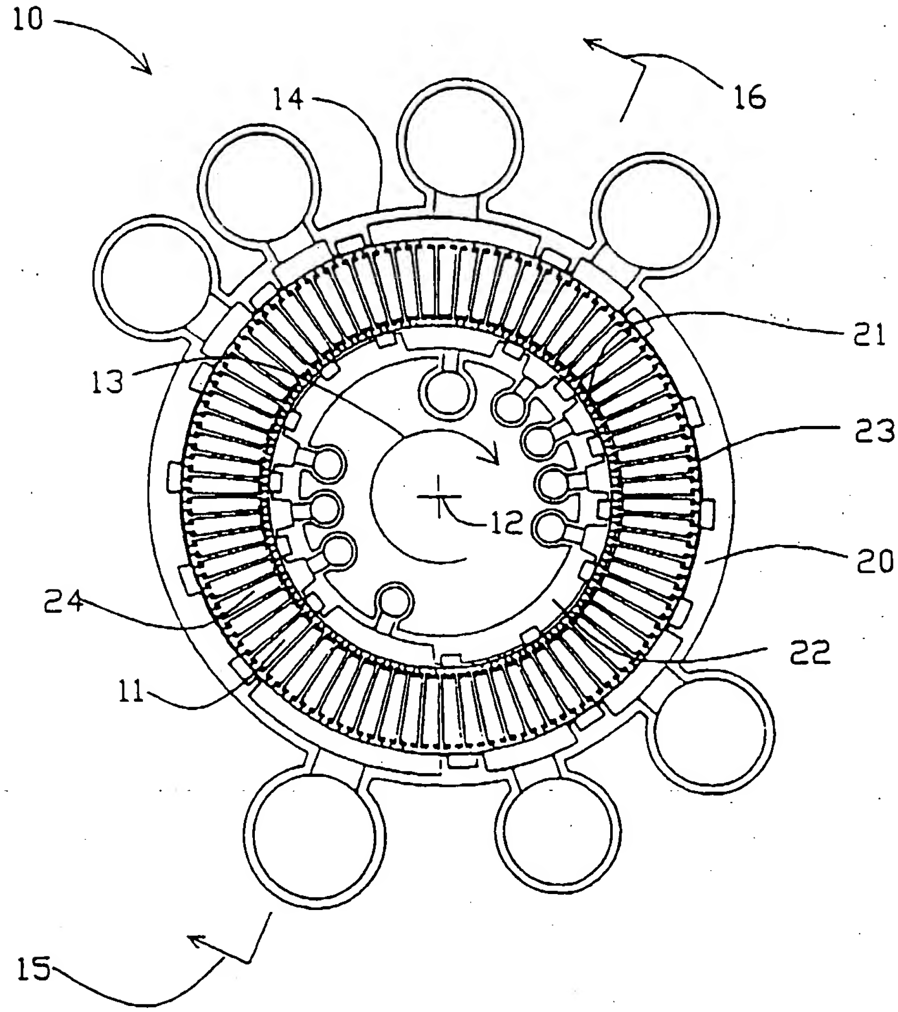


Fig. 1

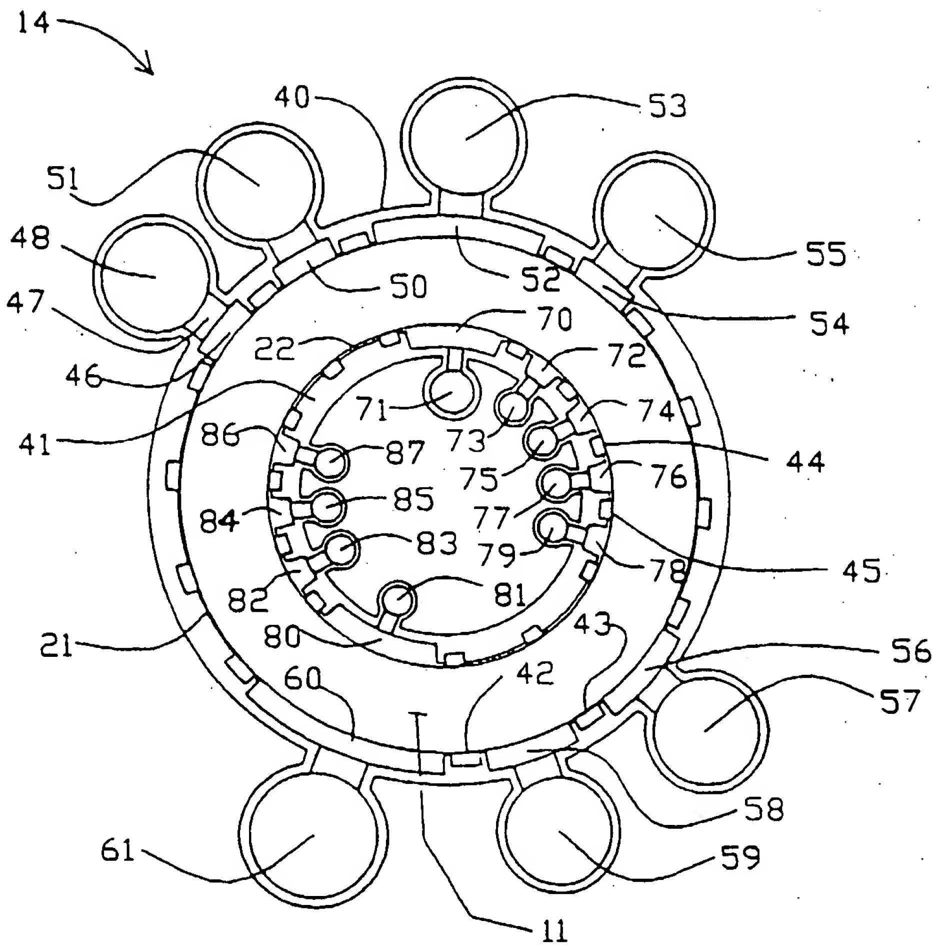

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Fig. 3

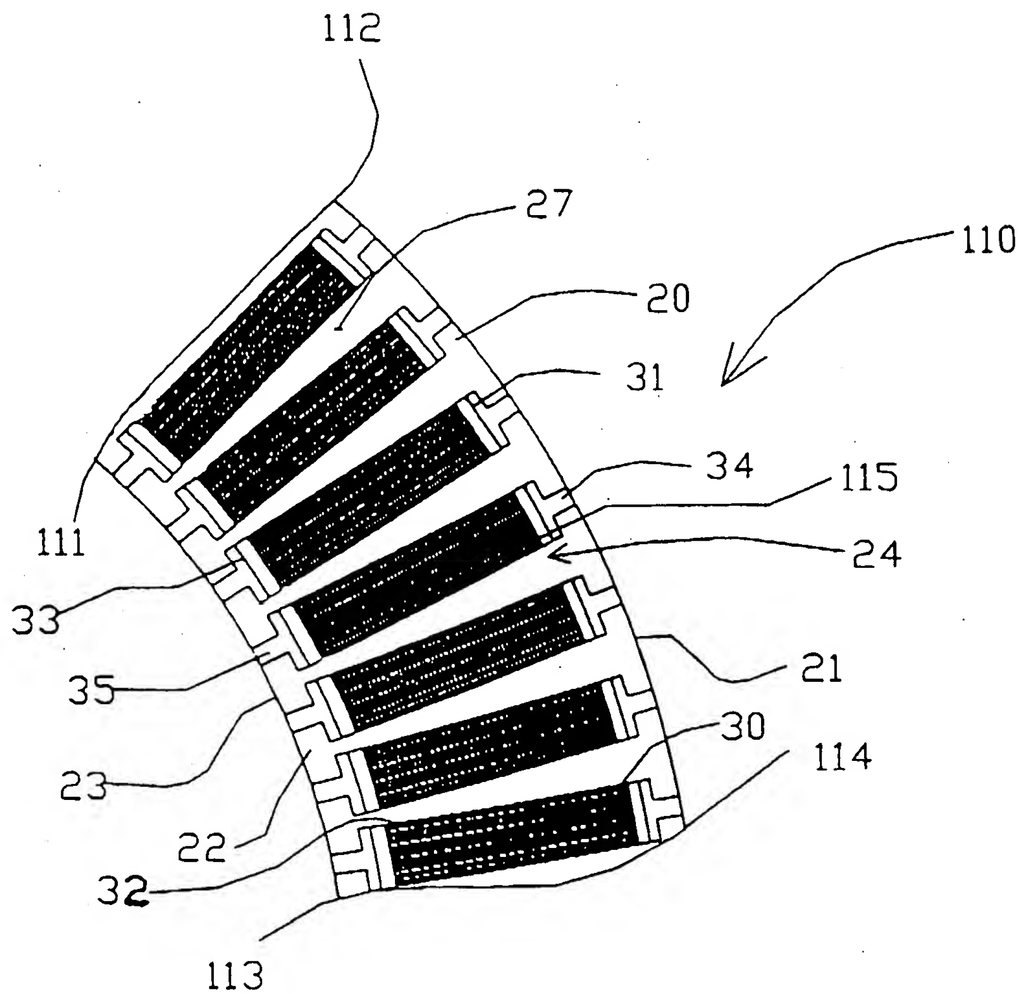

Fig. 5

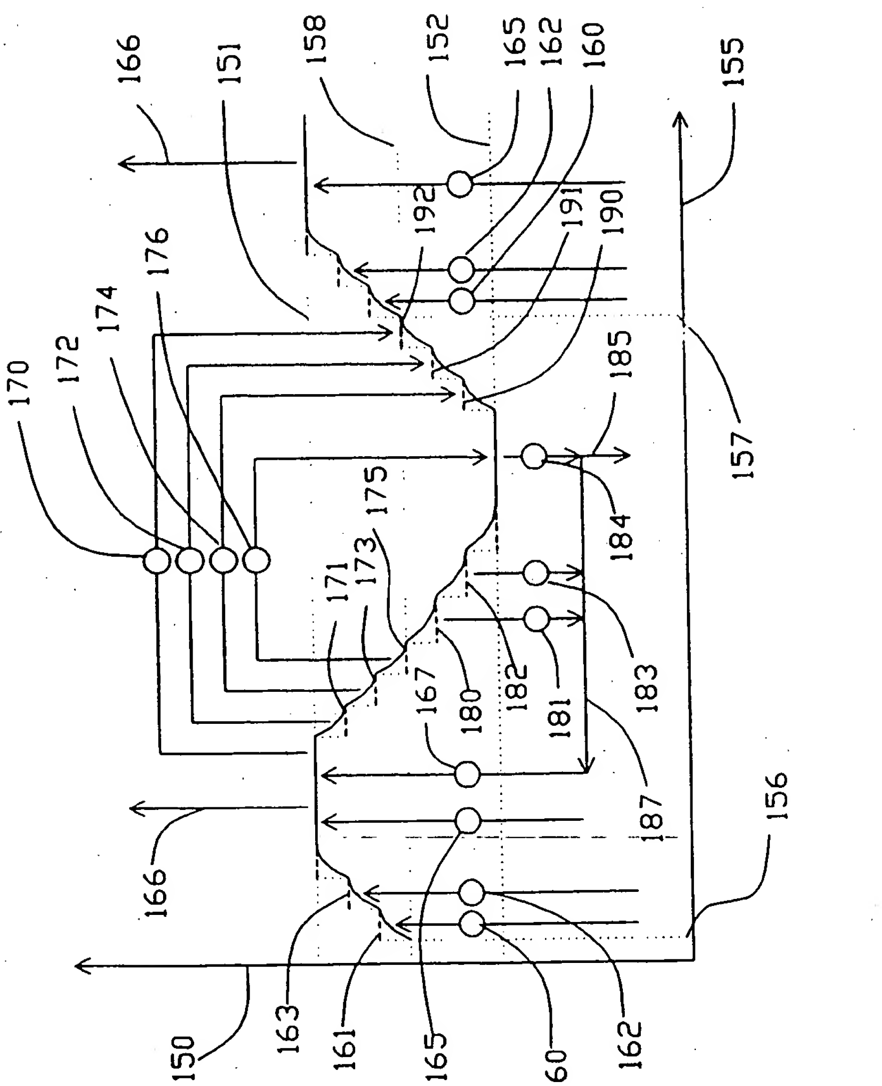
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Fig. 7

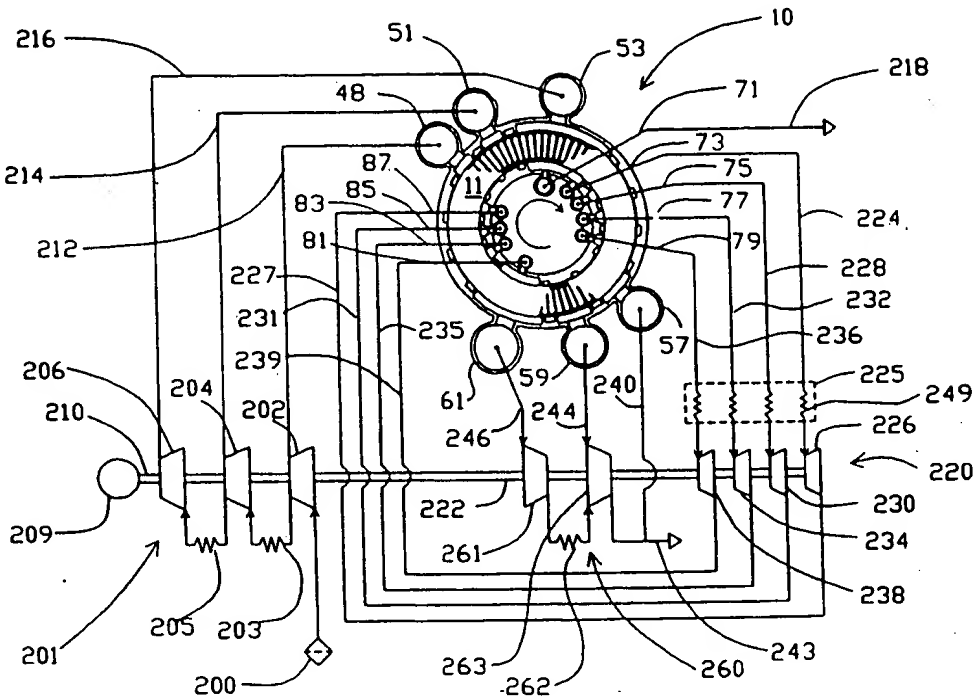
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Fig. 9

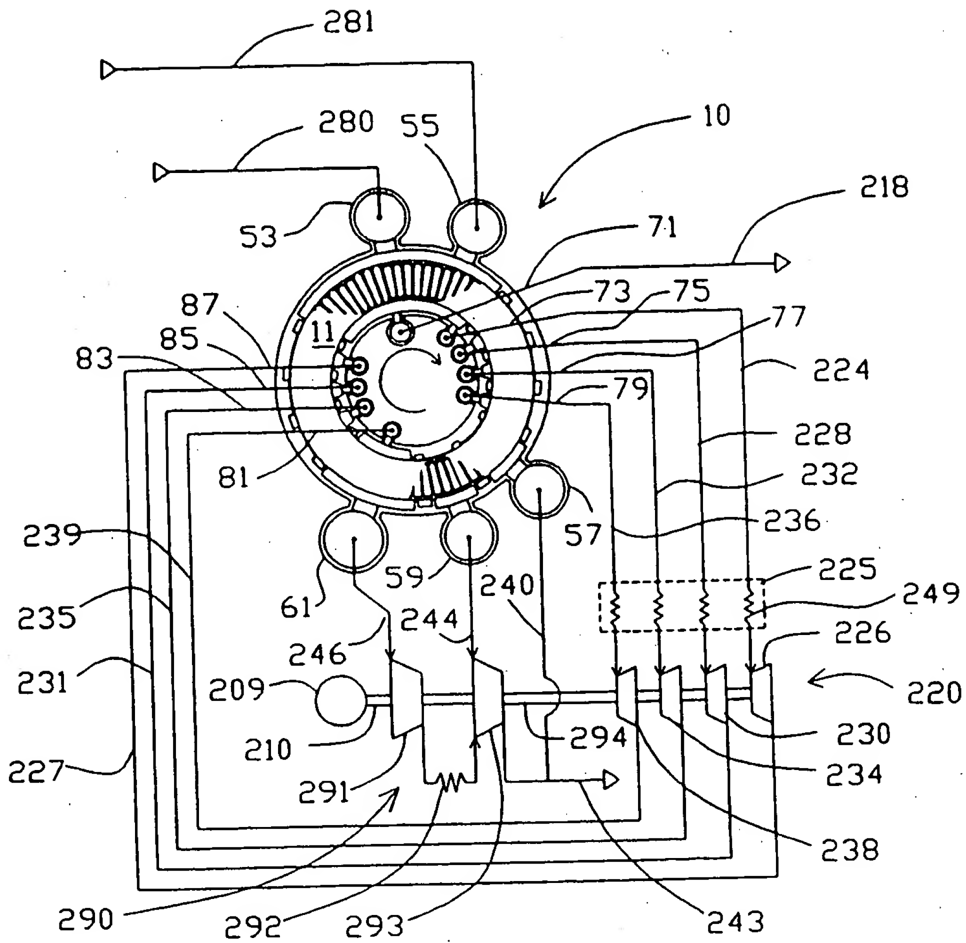
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Fig. 11

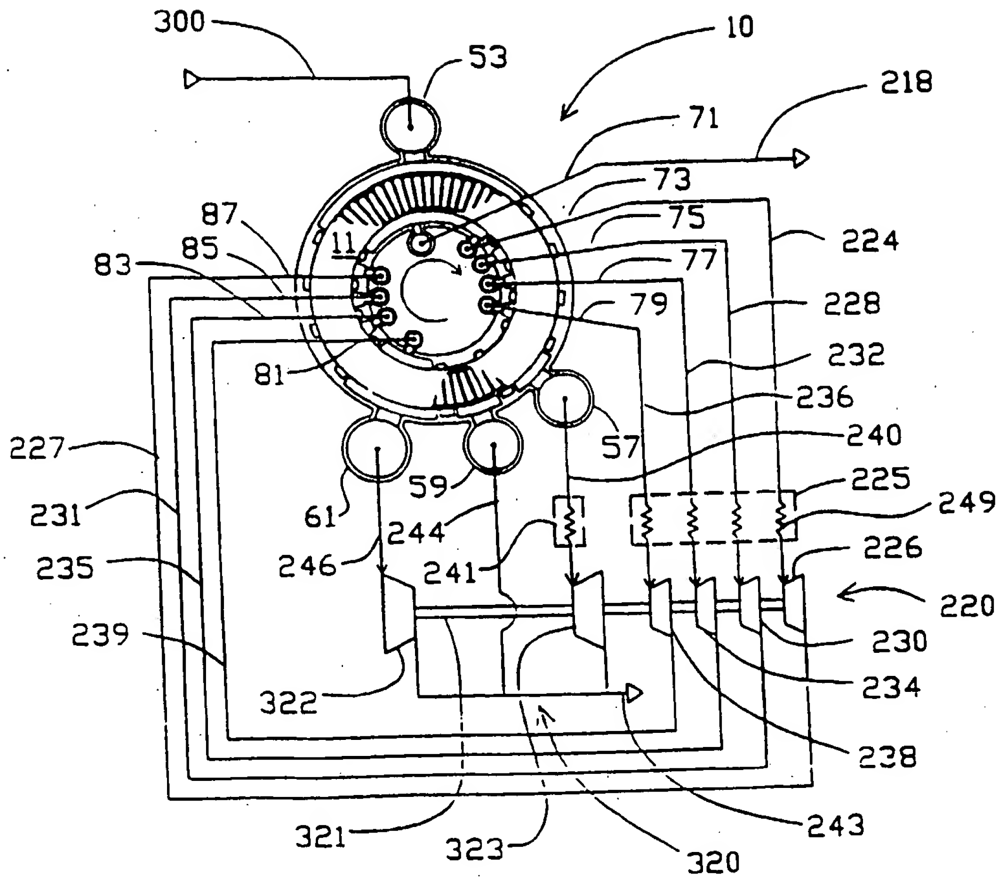
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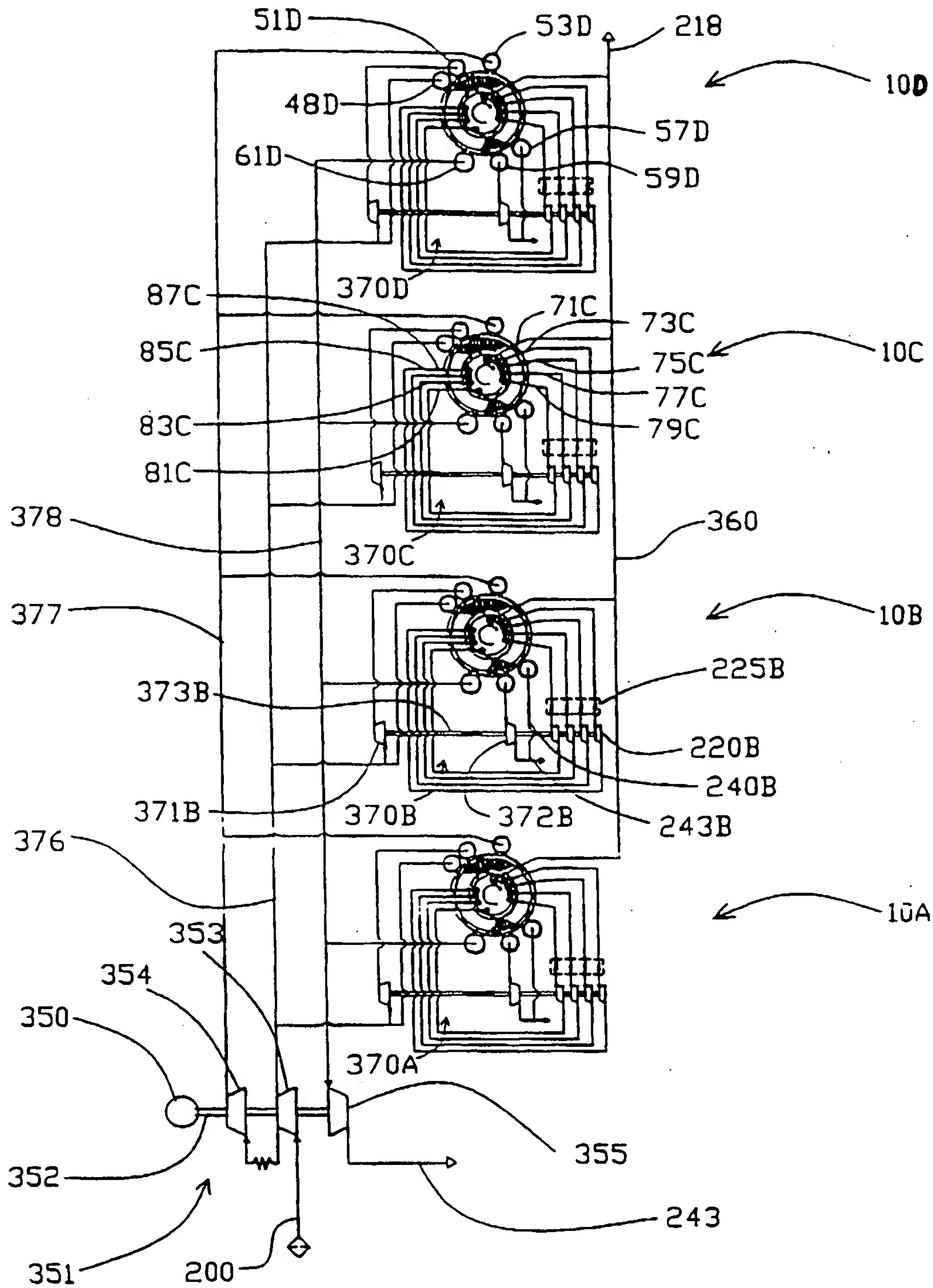
Fig. 13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373B
220B
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Fig. 15

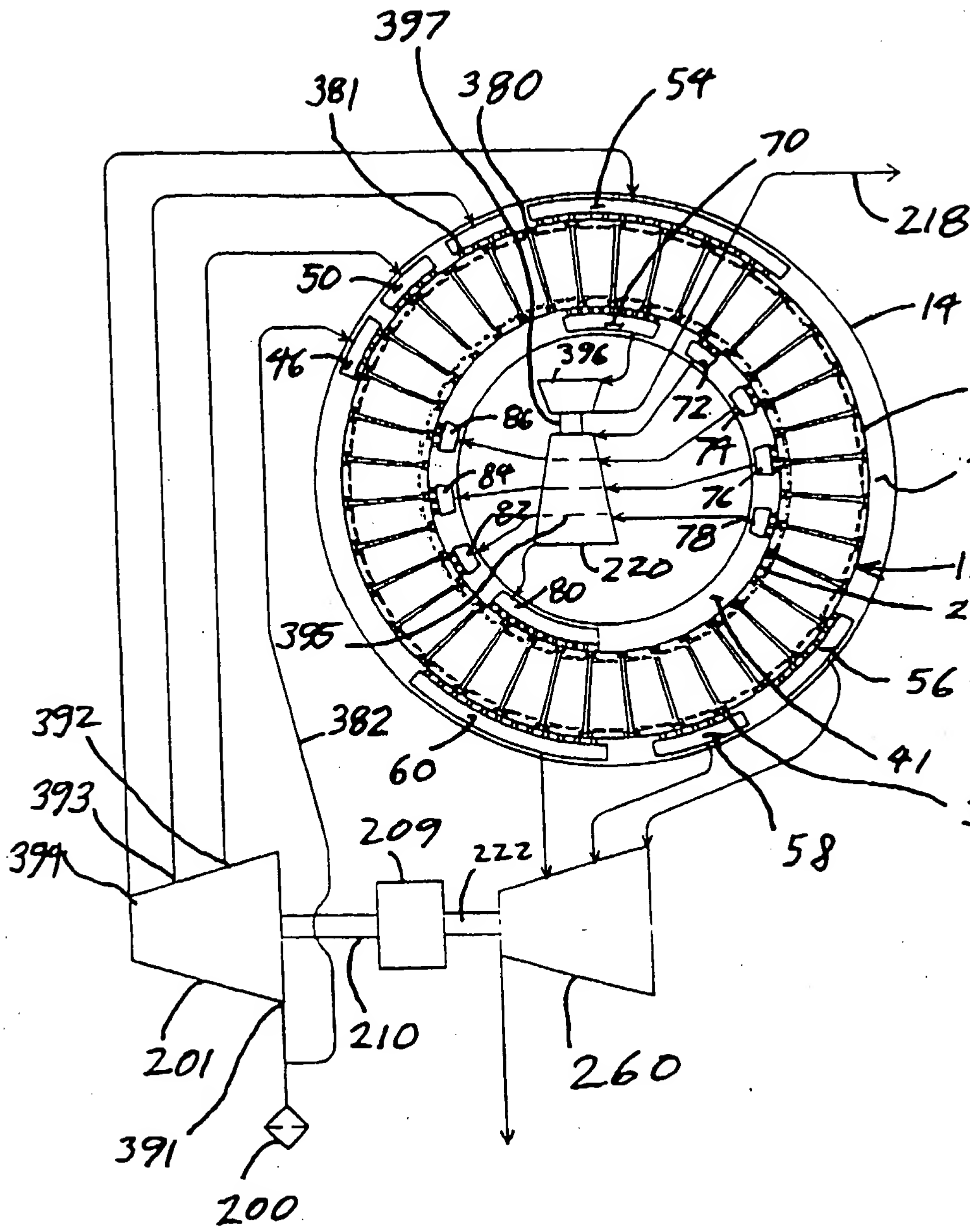
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FIG. 17

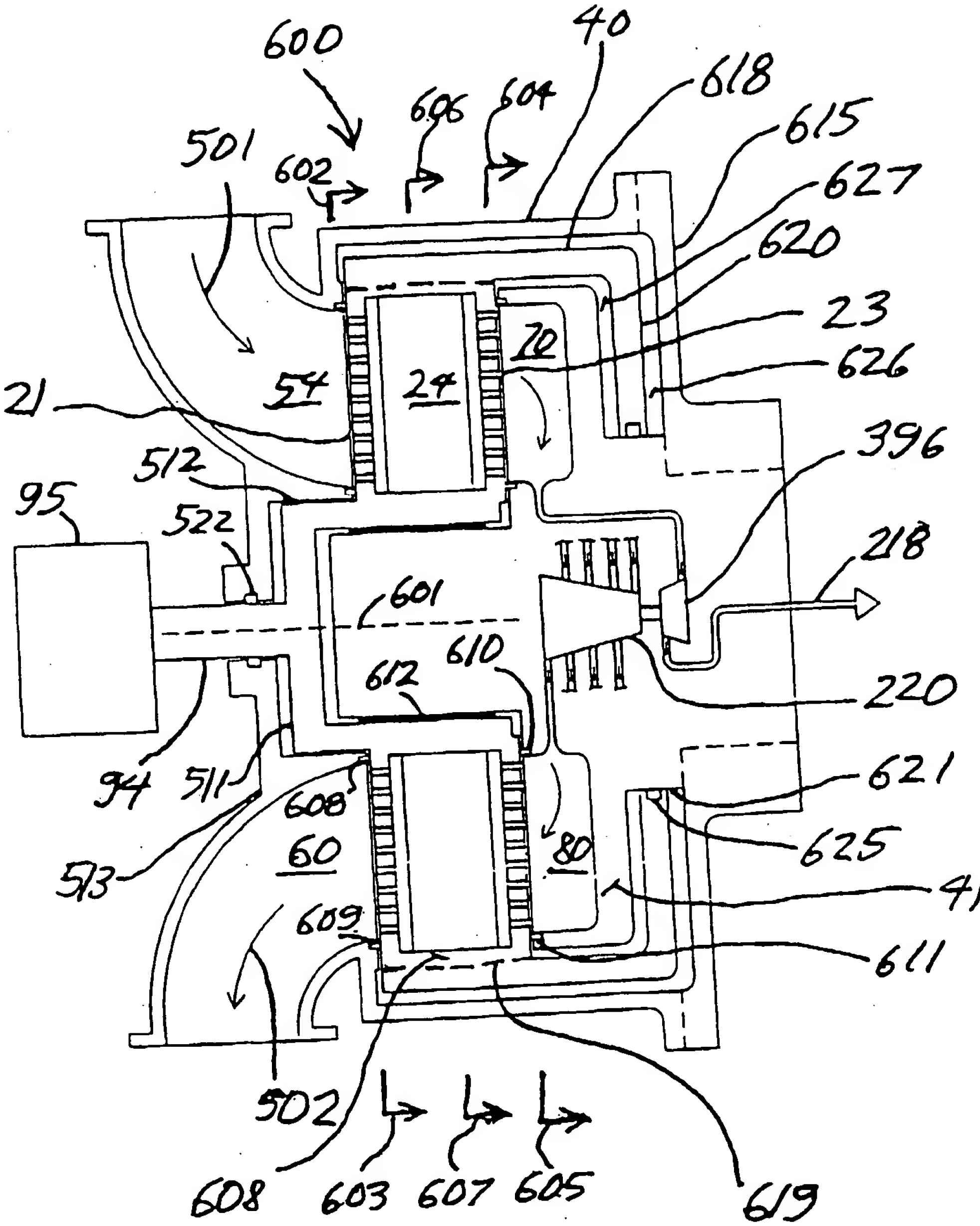
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FIG. 19

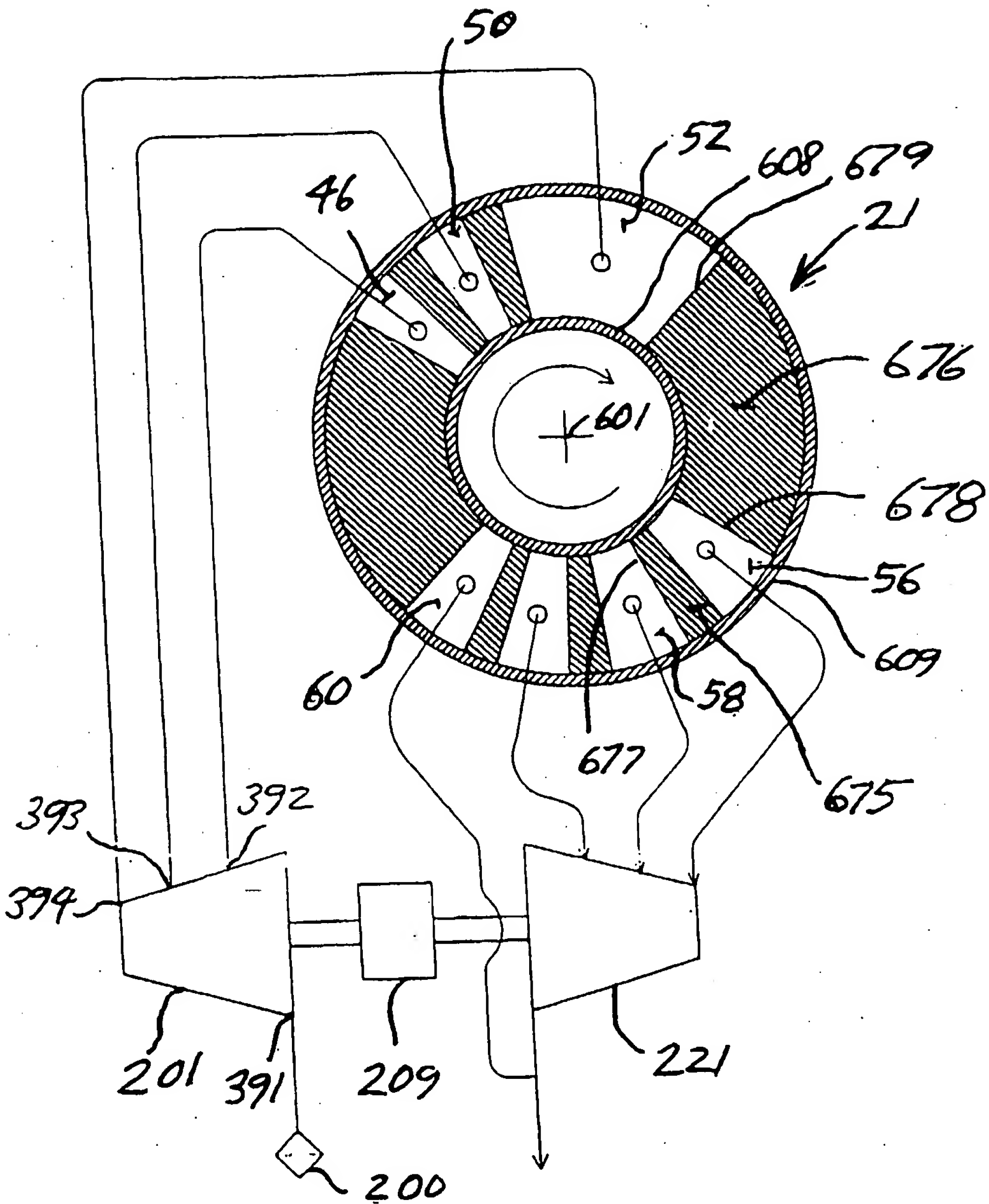
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FIG. 21

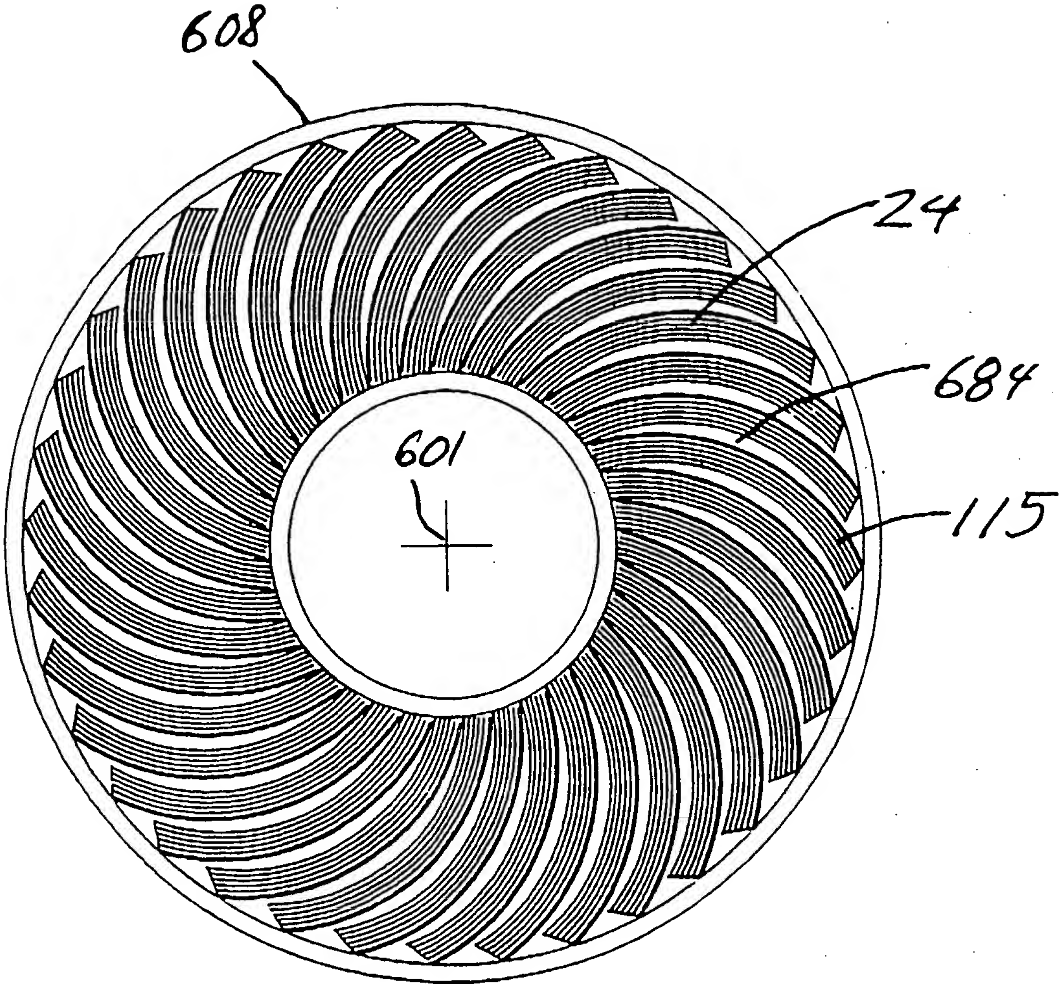

FIG. 23

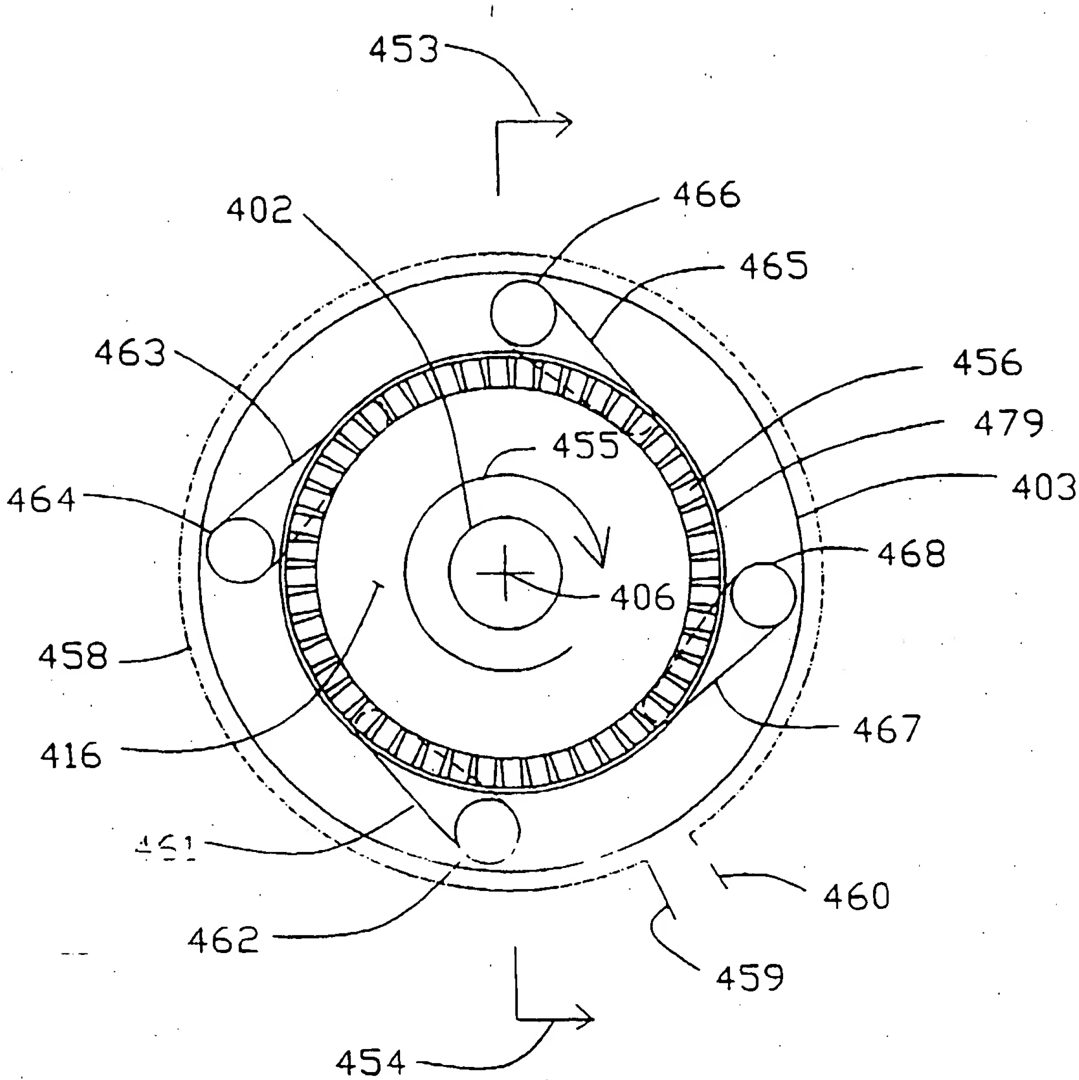
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Fig. 25

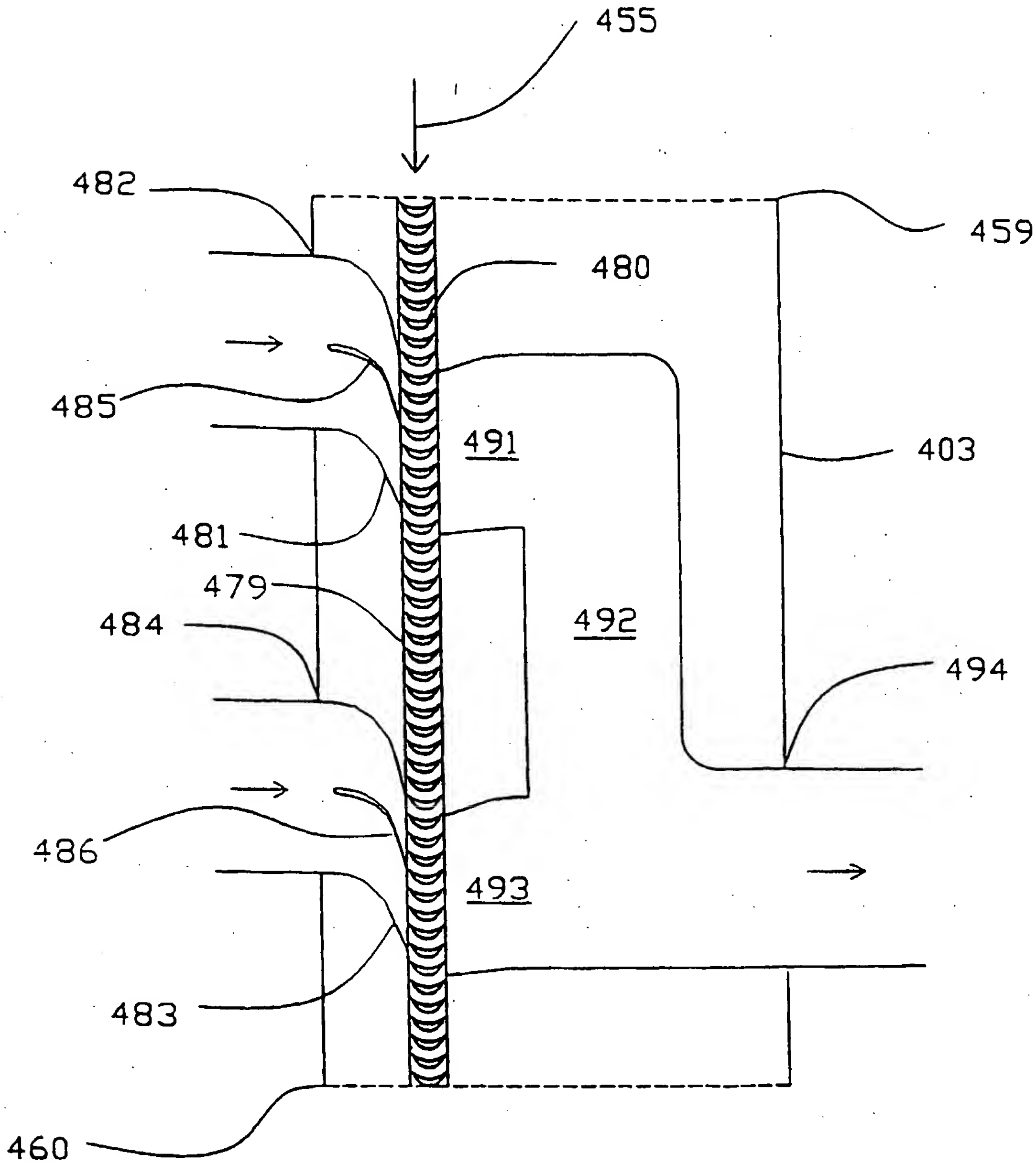

Fig. 27

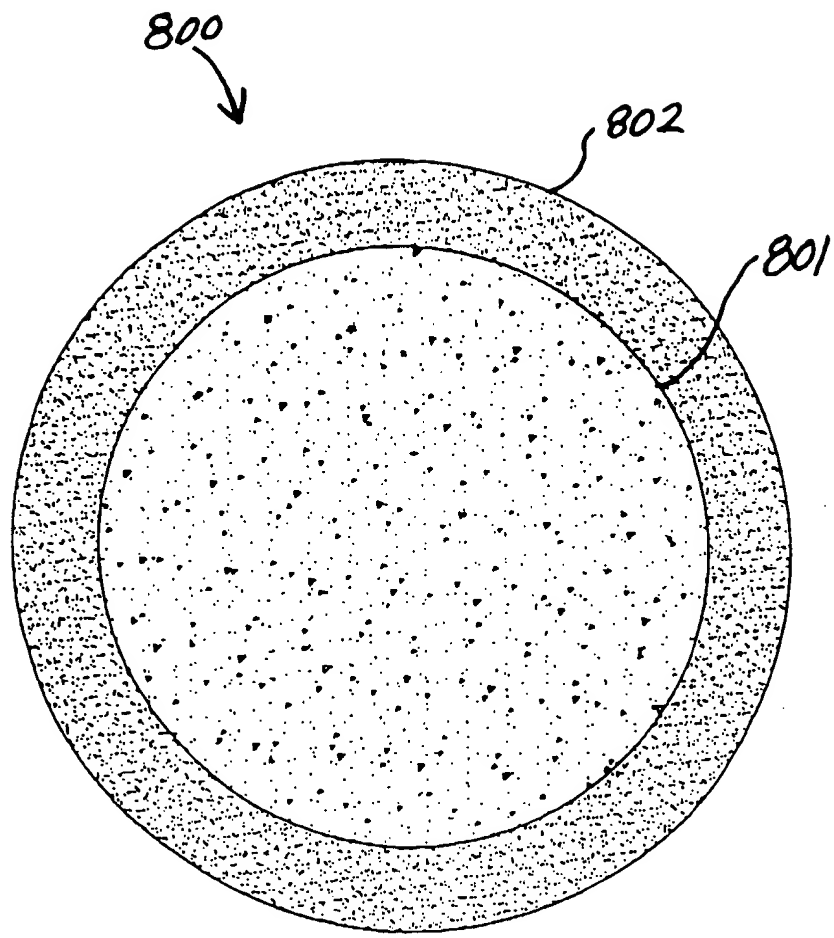

FIG. 29